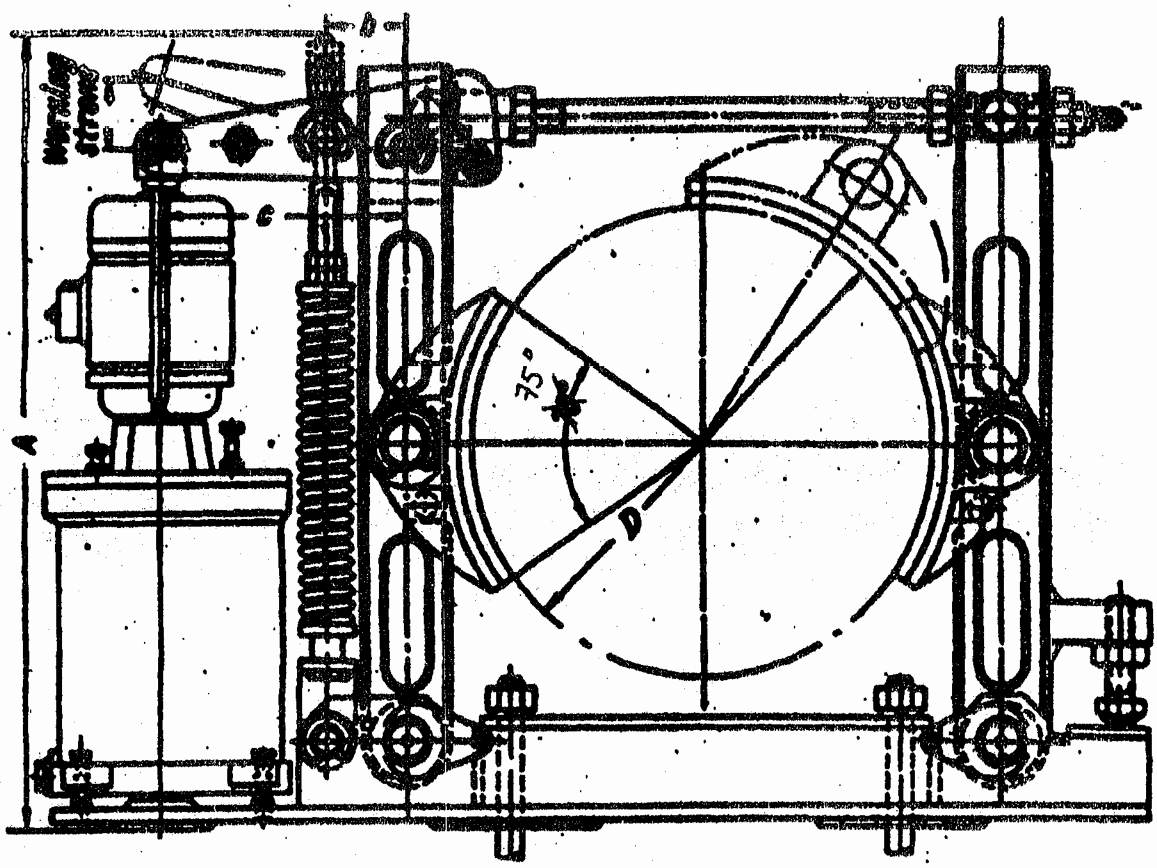


Normaly-On Drum Brake with External Shoes and an Electrohydralic Thrustor.

∫ Given Data:

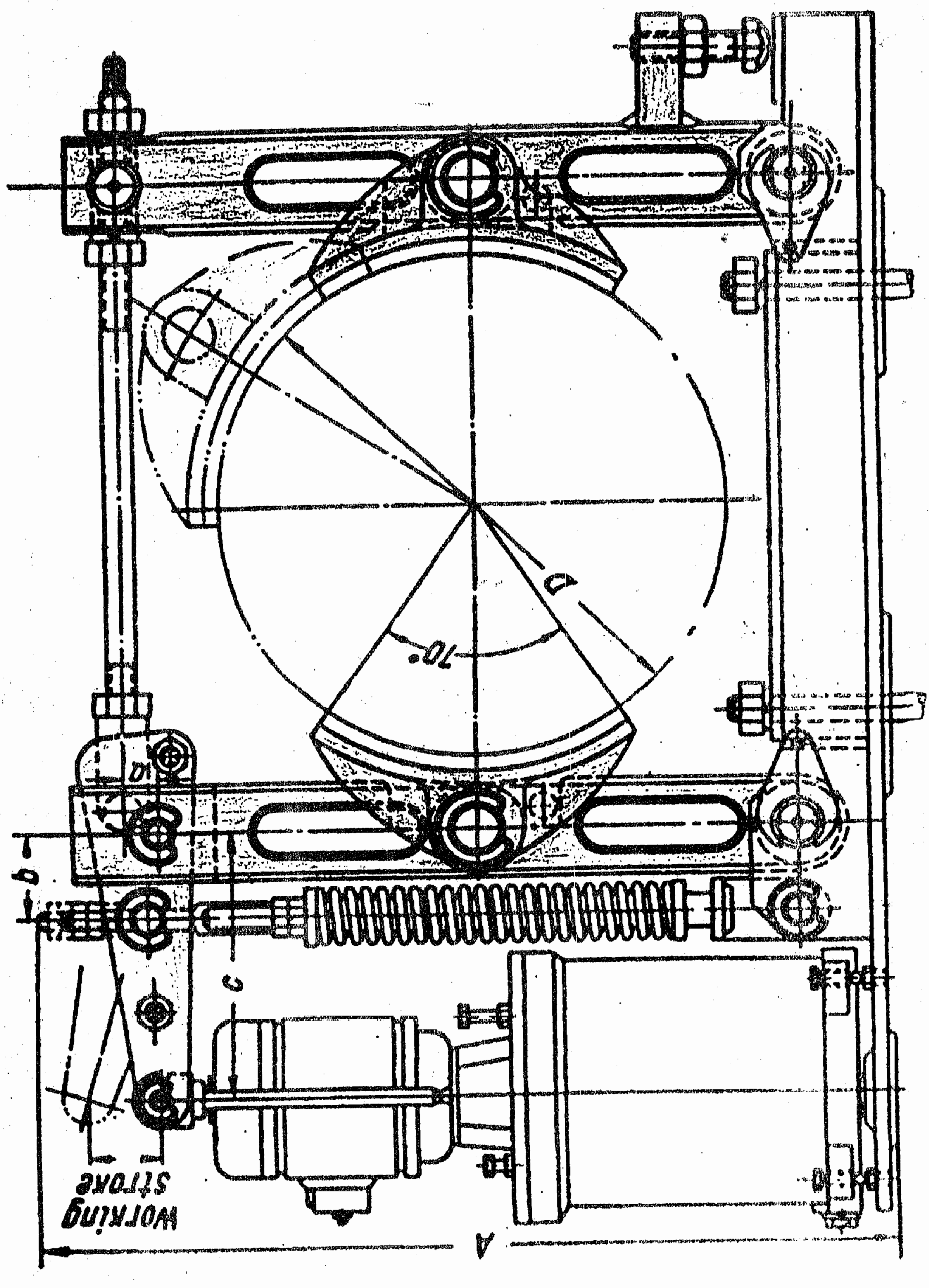
* Thrustor:
  - Max. force/max. stroke 300N/25 mm
  - Actuation time 0.2 s
  - Motor Power 110 W
  - Pressure 60 kPa
  - Bore/max. outside Diam. 90 mm/120 mm
  - Closed hight ≃ 300 mm

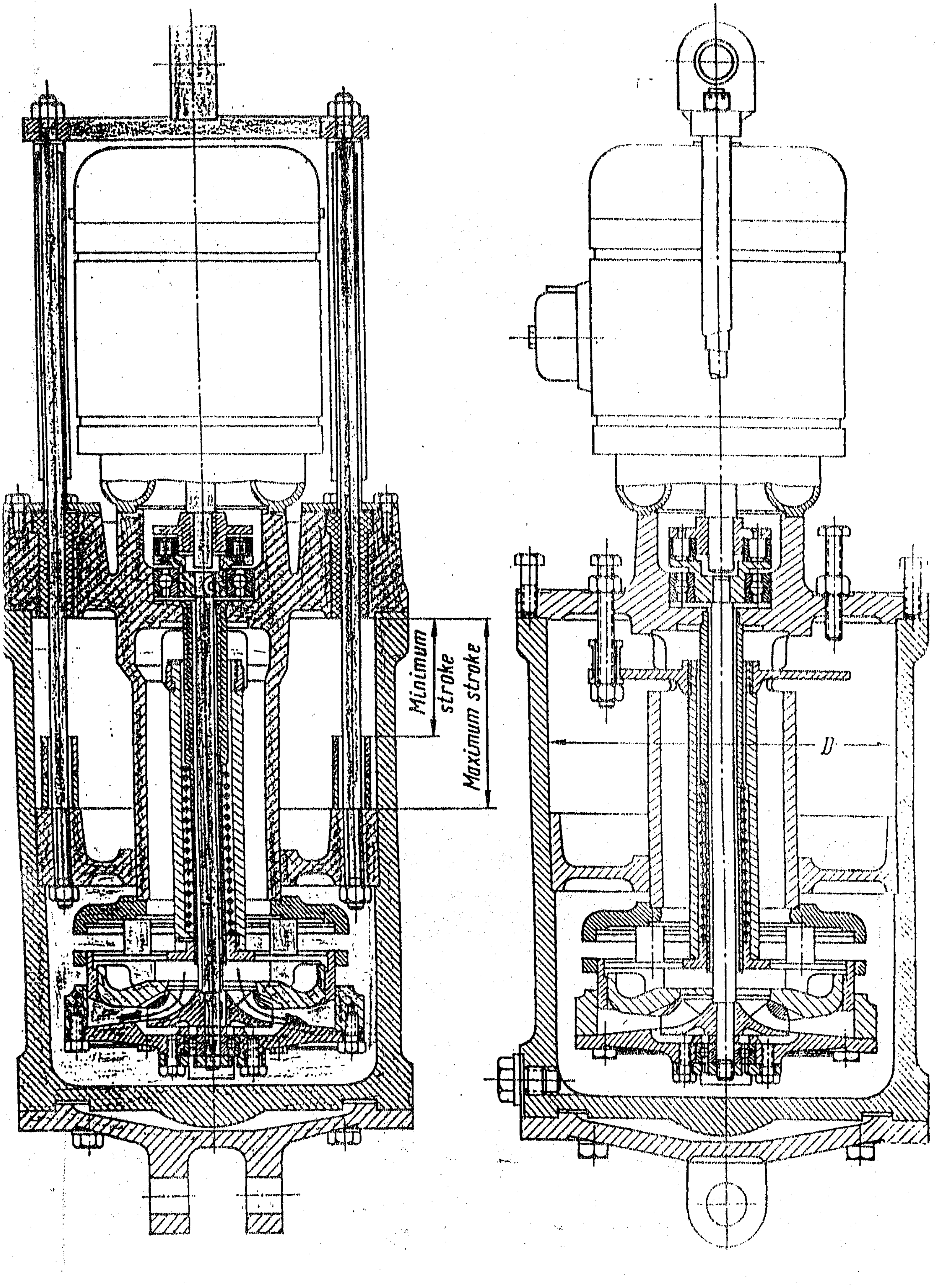
* Actuating Lever:
  - a/b/c = 22.5/45/135 mm

* Brake:
  - Drum diam/width 200/100 mm
  - Drum peripheral speed = 8 m/s
  - Safety margin = 4

∫ Required: If the Electrohydraulic thrusto is the suitable thrustor to disconnect the brake, you are asked to:

* Workout the forces and reactions on different elements.
* Make a complete stress analysis for the elements based on the loading diagrams and hence select the suitable materials.
* Construct the details of the joints necessary to realize the function
* What is the max. torque this brake can arrest ?


working stroke
A
D
70°
b
c



Electrohydraulic thrustor

Shoe Brake Projects:

I- Selection of appropriate thrustor size :-

1- Drum size = ✓ ; Shoe width = 0.3 – 0.4 drum Diam. ✓✓

2- Brake shoe angle = 75° ✓, Drum size = ✓ ; projected length = ✓✓

3- Projected length = ✓, width = ✓; area = ✓✓

4- Select lining material (table) ✓, allowable pressure = ✓, Service factor = 1.5 – 2 ✓; total reaction = ✓✓

5- Right-hand lever main dimensions ✓, reaction at intermediate point = ✓ ; tension in tie rod = ✓✓

6- Actuating lever dimensions ✓, $P_{actuator} = 0.0$, tension in tie rod = ✓ ; F_{spring} = ✓✓

7- Actuator lever dimensions = ✓, F_{spring} = ✓, $F_{rod} = 0$; $P_{actuator} \geqslant$ ✓✓

Hence Select the actuator.

~Tit

*How to enhance and adjust the calculated values for the available actuator sizes?

Either:

increase the arm length (and hence the stroke)

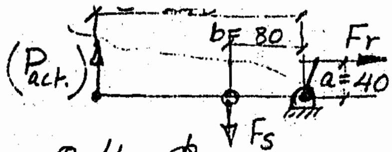
OR:

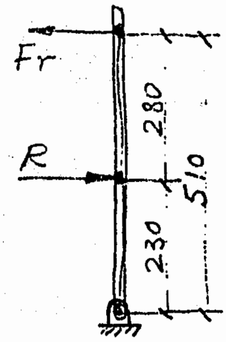
reduce projected area by reducing the shoe width (i.e. the reaction decreases, F_{rod} gets less, F_{spring} gets smaller and consequently the actuator force will decrease.)

In general: both approaches could be combined in an adequate way if each alone is not enough.

Next step:

Proceed like the solved example to get through the design fixing the material, the selected dimensions in the Elevation View and workout the limit load carrying capacity of this brake arrangement.





$\phi_{drum} = 400$ mm

shoe width $= 0.3 \rightarrow 0.4 \; \phi_{drum}$

$= 120 \rightarrow 160$ mm

Projected length of shoe $= 200$ mm

Selection of actuator:

Shoe width	120	140	150	160
Area (mm^2)	24,000	28,000	30,000	32,000
80% Area (mm^2)	19,200	22,400	24,000	25,600
Max. pressure (k.Pa)		690		
Service factor ($\gtrsim 1.5$)				
Adopted pressure (kPa)		450		
$R = p * Area$ (kN)	8.64	10.1	10.8	11.52
$F_r = (R * 230)/510$ (kN)	3.9	4.6	4.9	5.2
$F_s = Fr * \frac{40}{80}$ (kN)	1.85	2.3	2.45	2.6
* $P_{act.}$ to get $F_r = 0 = P_{act.\,min} = F_s * \frac{80}{240}$ (kN)	0.617	0.766	0.817	0.867
For $C = 300$, $b = 80$, $P_{act_{min}} = F_s * \frac{80}{300}$ (kN)	0.493	0.613	0.653	0.693

Select $P_{act\,max} = 600$ N, Max stroke of 75 mm

For stroke $= 70$ mm.:

$F_{s_{max}} = 600 \times \frac{300}{80}$	kN	2.25
$\delta_{spring} = 70 * \frac{80}{300}$	mm	$\simeq 19$
$F_{s_{max}} - F_s = \Delta F_s$	kN	0.4
K = Spring stiffness $= \frac{\Delta F_s}{\delta_{spring}}$	N/mm	21
Assembly $\delta_{spring} = \frac{F_s}{K_{spring}} = \frac{1850}{21}$	mm	88
δ total $= \delta_{spr.} + \delta_{ass.}$		107

$n = 20$ eff. Coils

Single spring: $K=\dfrac{G\,d}{8C^3n}$, $\tau=\dfrac{8PC}{\pi d^2}k$, $k=\dfrac{4C+2}{4C-3}$

index C	6	7	8
K (N/mm)		21	
δ assembly (mm)		88	
δ working (mm)		19	
n (–)		20	
$d=\dfrac{8KC^3n}{G}=\dfrac{8*21*20}{8*10^4}C^3$	9.072	14.406	21.504
$D_o=d(C+1)$	63.5	115.25	193.54
$k=\dfrac{4C+2}{4C-3}=\dfrac{26}{21}$	1.238	not suitable for the construction	
total clearance at P_{max} (mm)	23		
δ Free to Solid $=88+19+23$	130		
$P_{peak}=\delta * K$ (N)	2730		
$\tau=\dfrac{8PC}{\pi d^2}=\dfrac{8*2730*1.238}{\pi\,(9)^2}$	106	N/mm^2	

Material Music wire DIN 17223 Sheet 1, $\tau_{perm.}=675\ N/mm^2$

Solid length $d*(n+2)$	$\simeq$ 200	mm
Free length $=l_{solid}+\delta$	330	mm
Assembled length $=l_{free}-\delta_{ass.}$	242	mm
Dm	55	mm
Do	64	mm
Di	46	mm

Use steel pipe 3" $(d/D)\simeq 78/89$ mm.

& a steel pipe $1\frac{1}{4}$" (33/42 mm) as an inside guide

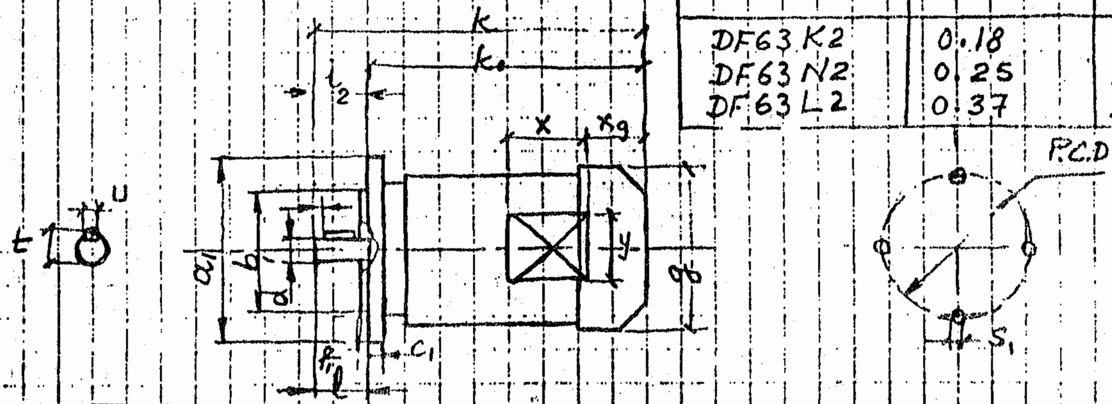
since: $(3D_o=3*64=192\text{ mm}<330\text{ mm})$

* Experimental Characteristics of Electrohydraulic Thrustor

Item	Characteristics	Units	Type of Thrustor and Values				
			T-I-55/75		T-II-55/135		
1	Rated force	N	600		1600		
2	Maximum stroke	mm	55	75	55	85	135
3	Duration of max. hoisting	s	0.5	0.73	0.65	1.0	1.5
4	Rated work	N.m	33	48	96	144	224
5	Speed of motor	r.p.m	2840 (3000)				
6	Motor rating	kW	0.144		0.326		
7	Oil pressure	MPa	0.031		0.03		
8	Piston diam.	mm	170		270		
9	Pump delivery	l/s	2.16		4.93		
10	Approximate wt. (dry)	N	400		700		

** Data Sheet for Fraction AC-Squirrel-Cage Motors

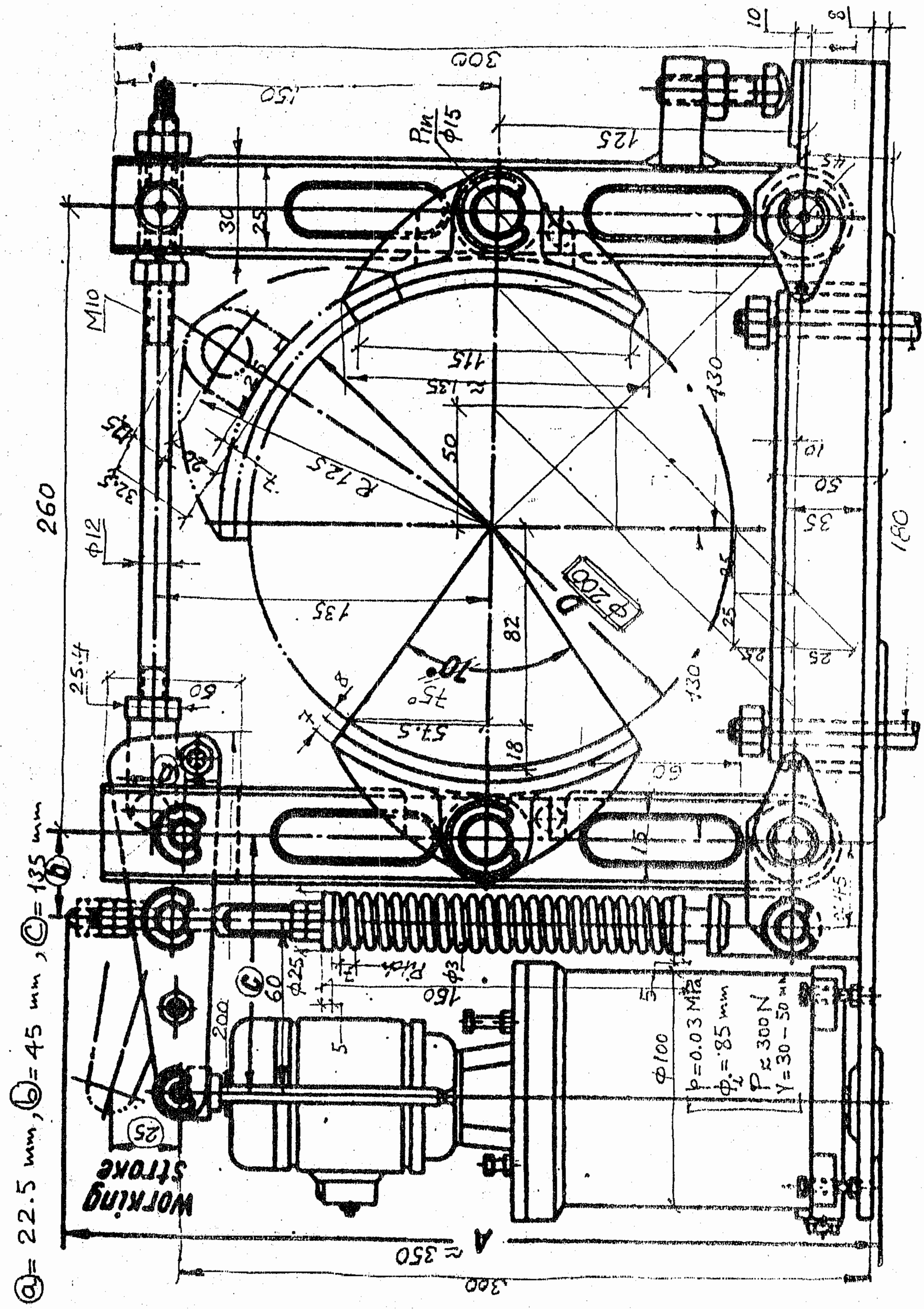
Frame type	Power kW	R.P.M.
DF63 K2	0.18	2720
DF63 N2	0.25	2660
DF63 L2	0.37	2650

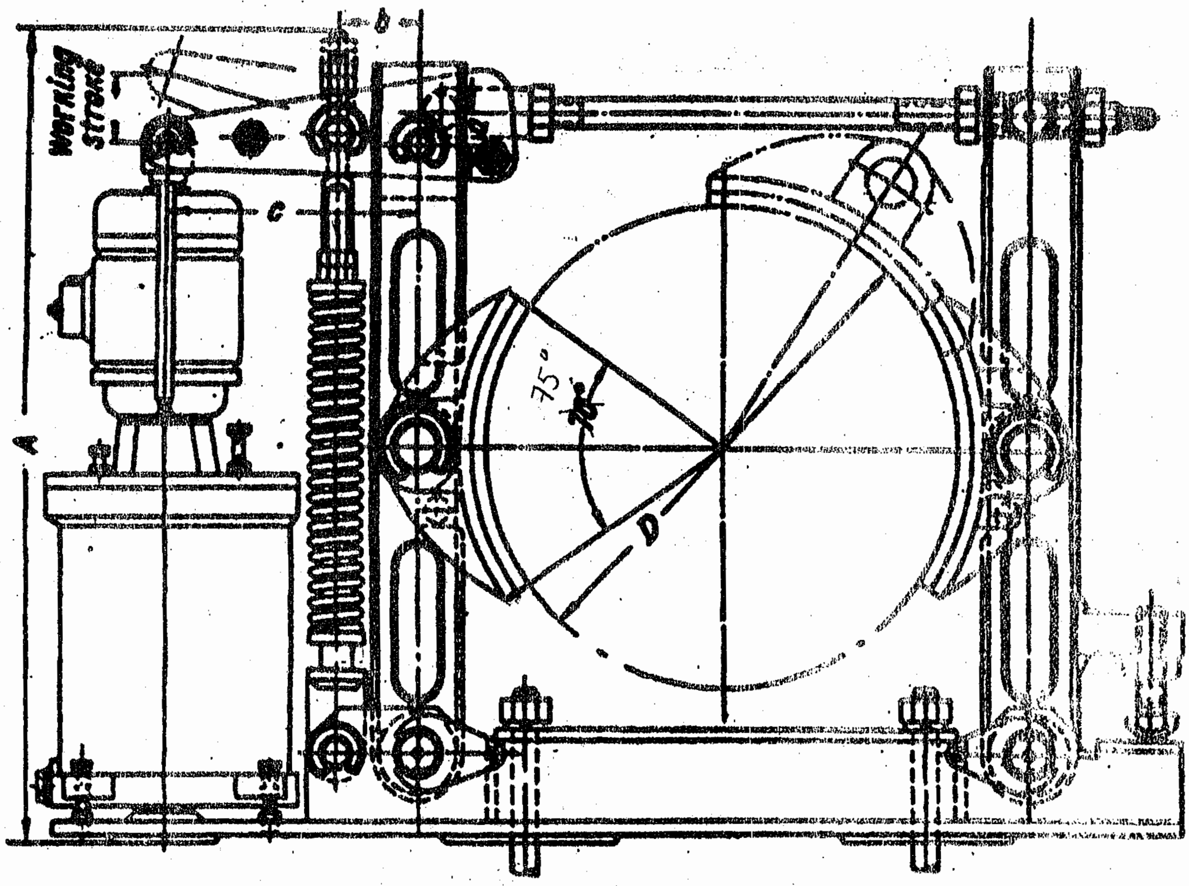


3000 r.p.m. Frame	a_1	b_1	c_1	f_1	i_2	k	k_o	S_1	x_g	x / y	d / l	t / u	P.C.D
DF63..	140	95	9	3	23	215	192	9	54	100 / 100	11 / 23	12.6 / 4	115

* Materials Handling Equipment, N. RUDENKO, MIR Pub. MOSCOW

** SEW Eurodrive, TÜV certified, Geared Motors Catalogue, 1993


ⓐ = 22.5 mm, ⓑ = 45 mm, ⓒ = 135 mm
Working stroke
Pin ϕ15
M10
R 125
ϕ200
D
ϕ12
ϕ25
ϕ3
ϕ100
p = 0.03 MPa
P ≈ 300 N
A ≈ 350
300
260
180



Normaly-On Drum Brake with External Shoes and an Electrohydralic Thrustor.

Given Data;

*** Thrustor:**

- Max. force/max. stroke 300 N/30 mm
- Actuation time 0.2 s
- Motor Power 110 W
- Pressure 60 kPa
- Bore/max. outside Diam. 90 mm/120 mm
- Closed hight ≅ 300 mm

*** Actuating Lever;**

- a/b/c = 22.5/45/135 mm

*** Brake:**

- Drum diam/width 200/100 mm
- Drum peripheral speed = 8 m/s
- Safety margin = 4

Required:

If the Electrohydraulic thrustor is the suitable thrustor to disconnect the brake, you are asked to:

* Workout the forces and reactions on different elements.

* Make a complete stress analysis for the elements based on the loading diagrams and hence select the suitable materials.

* Construct the details of the joints necessary to realize the function

* What is the max. torque this brake can arrest?

Thrustor : (Electrohydraulic)

$P_{act} = 300$ N = max. output of the thrustor

Stroke = 30 mm (min.), 50 mm (max.)

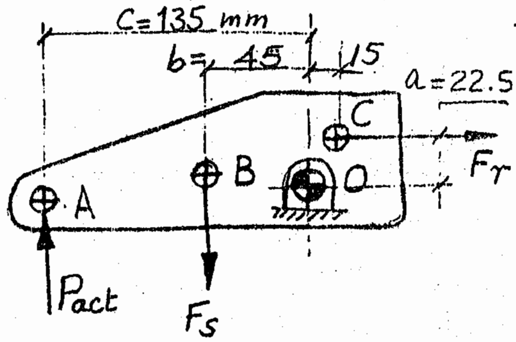
$P_{act} = 300$ N

Stroke = 30 mm

F_r = tie rod force

F_{s_1} = Spring force when $P_{act} = 0$

F_s = max. spring force at the end of the lever stroke.



When the brake is just released, $F_r = 0$

Moment about O = 0.0

$P_{act} * C = F_s * b \quad \therefore F_s = \dfrac{300*135}{45} = 900$ N

$F_s = 900$ N

Since P_{act} is the max. thrustor force, then the $F_s = 900$ N is the spring force when fully deflected at the end of thrustor stroke.

Spring dimensional chs:

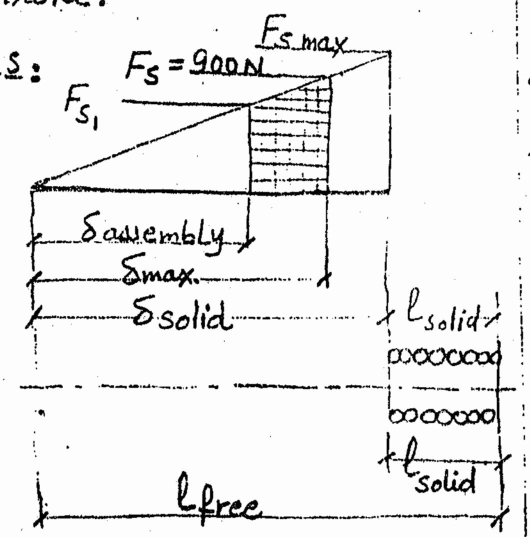
$n_{eff} = 20$

$D_o = 25$ mm

$d = 3$ mm

$\therefore D_m = 22$ mm

$C = \dfrac{22}{3} = 7.33$



Spring assume dims.:
$n_{eff} = 20$
$D_o = 25$ mm
$d = 3$ mm
Square ground ends.

$$k = \frac{4C+2}{4C-3} = \frac{31.32}{26.32} = 1.2$$

$k = 1.2$

Spring stiffness = K

$$K = \frac{G\,d}{8C^3 n} = \frac{8*10^4*3}{8(7.33)^3*20} = 3.8 \text{ N/mm}$$

$K = 3.8$ N/mm

$$\delta_{work.} = \delta_{max} - \delta_{ass} = \text{thrustor stroke} * \frac{45}{135} = 10 \text{ mm}$$

$\delta_{wor} = 10$ mm

$\therefore F_{s_1} = 900 - (3.8*10) = 862$ N

$l_{solid} = n_{eff} * d + 2d \longrightarrow$ Square ground ends

$\therefore\ l_{solid} = 22 * 3 = 66$ mm

$\delta_{ass} = F_{S_1} / K = 862/3.8 = 227$ mm

Since $l_{ass} = 150$ mm and, $pitch_{ass} = 7$

then, $l_{free} = 150 + 227 = 377 \text{ mm} = n * pitch_{free} + 2d$

i.e. $pitch_{free} = (377-6)/20 = \frac{371}{20} = 18.5$ mm

δ to solid sping $= l_{free} - l_{solid} = 377 - 66$

$= 311$ mm (<u>not practical</u>)

$\tau \geq \frac{8 P_{max} C}{\pi d^2} k$ where

$F_{S\,max} = P_{max} = 311 * 3.8 = 1182$ N

$\tau_{perm} \geq \frac{8 * 1182 * 7.33 * 1.2}{\pi (9)} \geq 2941$ N/mm^2

τ_{perm} is too high.

$\tau_{work} = 2941 * \frac{900}{1132} = 2240$ N/mm^2 $\longrightarrow$ <u>Too high</u>

Three trials for $d = 3.5, 4, 4.5$

Item	3	3.5	4	4.5
$C = \frac{25-d}{d}$	7.33	6.14	5.25	4.56
$k = \frac{4C+2}{4C-3}$	1.2	1.23	1.28	1.33
$K = \frac{G d}{8C^3 n} = \frac{10^4 d}{20 C^3}$ [N/mm]	3.8	7.57	13.8	$\boxed{23.7}$
$l_{solid} = (n+2)d$ [mm]	66	77	88	99
$\boxed{F_{S_1} = 900 - 10K}$ [N]	862	824	762	$\boxed{663}$
$\boxed{\delta_{ass} = F_{S_1}/K}$ [mm]	227	109	55	$\boxed{28}$
$l_{free} = 150 + \delta_{ass}$ [mm]	377	259	205	178
$\delta_{solid} = l_{free} - l_{solid}$	311	182	113	79
$P_{max} = F_{S\,max} = K\,\delta_{solid}$ [N]	1182	1378	1560	1872
$\tau = \frac{8 P_{max} C}{\pi d^2} \cdot k$	2941	2163	1668	1427
$\tau_{Working}$ [N/mm^2]	2240	1413	962	$\boxed{686}$
Material №	-		-	6

<u>Appropriate Spe</u>

$K = 23.7$ N/m

Assembly force 663 N

$\delta_{ass} = 28$ mm

$l_{free} = 178$ mm

$\tau_W < 860$ N/mm

oil Tempered Cr. Si

A) Force Analysis:

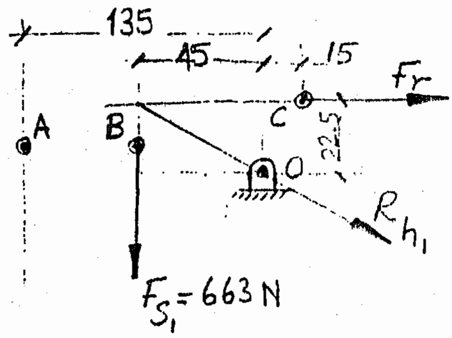
Forces in holding brake elements: (internal forces)

1) Actuating lever:

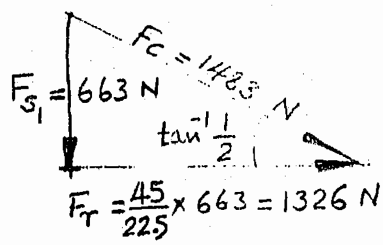
Based on $F_{S_1} = 663$ N,

$\therefore F_r = 1326$ N along the top tie rod

$F_{S_1} = 663$ N
$F_r = 1326$ N



Force Polygon.



2) Shoes Carrying levers:

Mts @ O_1 : [L.H. Lever]

$$125 * R = (125+120) * 1326$$

$\therefore R_L \equiv 2600$ N

μ = Coefficient of friction

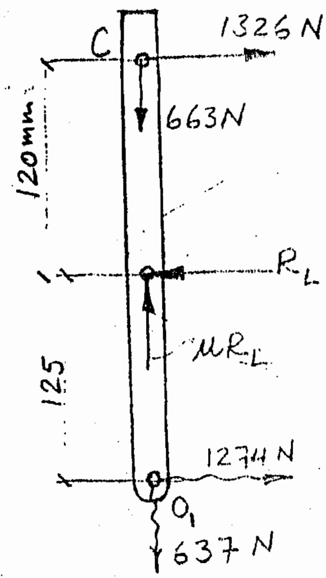
$\therefore \mu = 0.5$

$\therefore \mu R_L = 1300$ N

Reaction at O_1 :

$$R_{O_1 x} = 2600 - 1326 = 1274 \text{ N}$$

$$R_{O_1 y} = 1300 - 663 = 637 \text{ N}$$



Mts @ O_2 : [R.H. Lever]

$$R * 125 = 1326 * 260$$

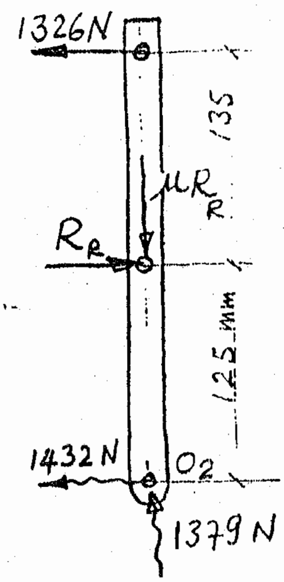
$\therefore R_R = 2758$ N

$\mu R_R = 1379$ N

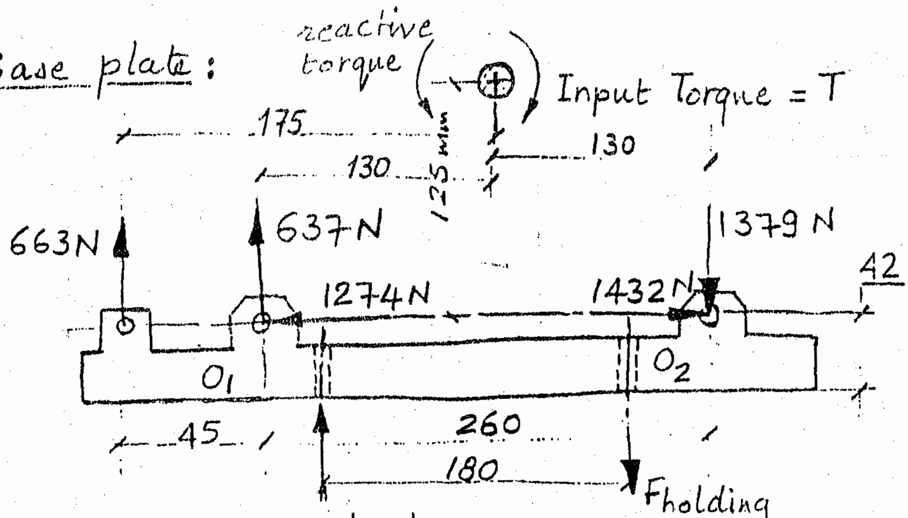
Reactions at O_2 :

$$R_{O_2 x} \cong 2758 - 1326 = 1432 \text{ N}$$

$$R_{O_2 y} = 1379 \text{ N}$$



3) Base plate :



Input torque to the brake assembly :

$$T_{work} = -\left[(1432-1274)*125\right]+\left[(1379+637)*130\right]+\left[663*175\right]$$

$$= -19750+262080+110775 = 353105 \text{ N.mm}$$

$$= 0.353 \text{ kN.m}$$

$T_{work} = 0.353$ kN.m.

4) Holding down bolts : (Two bolts)

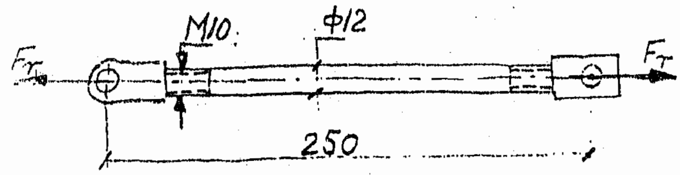
$$F_{holding} = \frac{353105}{180} = 1962 \text{ N}$$

Holding down force in each bolt = 1962 N

5) Tie rod (coupler) :

$F_r = 1326$ N

$F_r = 1326$ N



The rod construction details should allow for fine adjustment of brake clearance.

Stresses in different members:

1) Right-hand lever:

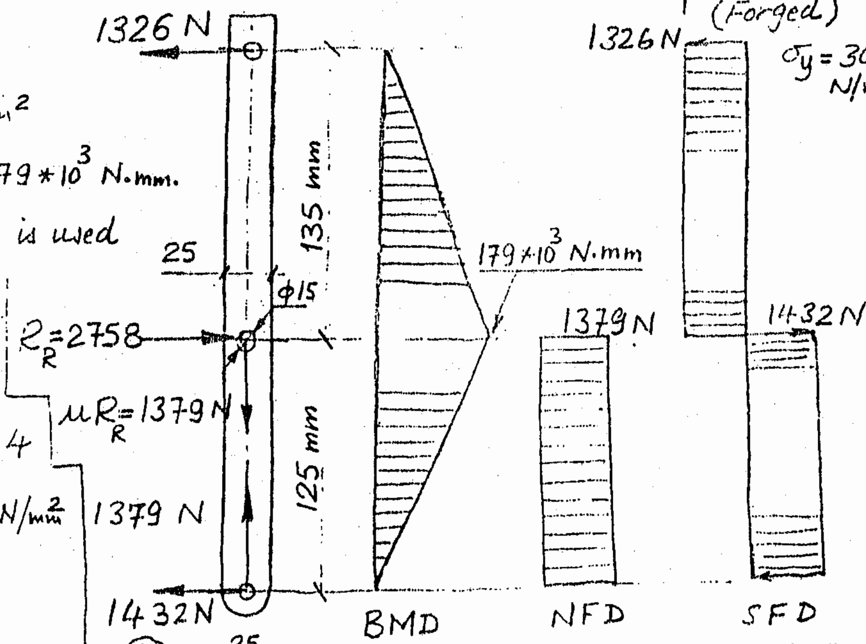
Material : C15

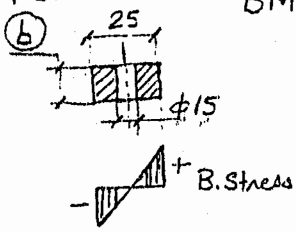
$\sigma_y = 300 \ N/mm^2$

$B.M. = 2758 * 125 = 179 * 10^3 \ N.mm.$

Since the element is used in lifting equipment, the margin of Safety should be as high as 4

$\therefore \sigma_{design} = \frac{300}{4} = 75 \ N/mm^2$





$\sigma_{design} = M/Z$

$\therefore Z = \frac{M}{\sigma_{design}} = \frac{179 * 10^3}{75}$

$= 2387 \ mm^3$

$Y = \frac{25}{2} = 12.5 \ mm$

$I = ZY = 12.5 * 2387 \cong 30{,}000 \ mm^4$

$I = \frac{b}{12}(25^3 - 15^3) = b(12250)/12 = 30{,}000 \ mm^4$

$\therefore b = 29$ mm taken 30 mm

Material: St C15 (Forged)

$\sigma_y = 300 \ N/mm^2$

Safety margin $= 4$

$\sigma_{design} = 75 \ N/mm^2$

$h = 25$ mm

$Y = 12.5$ mm

$I = 30{,}000 \ mm^4$

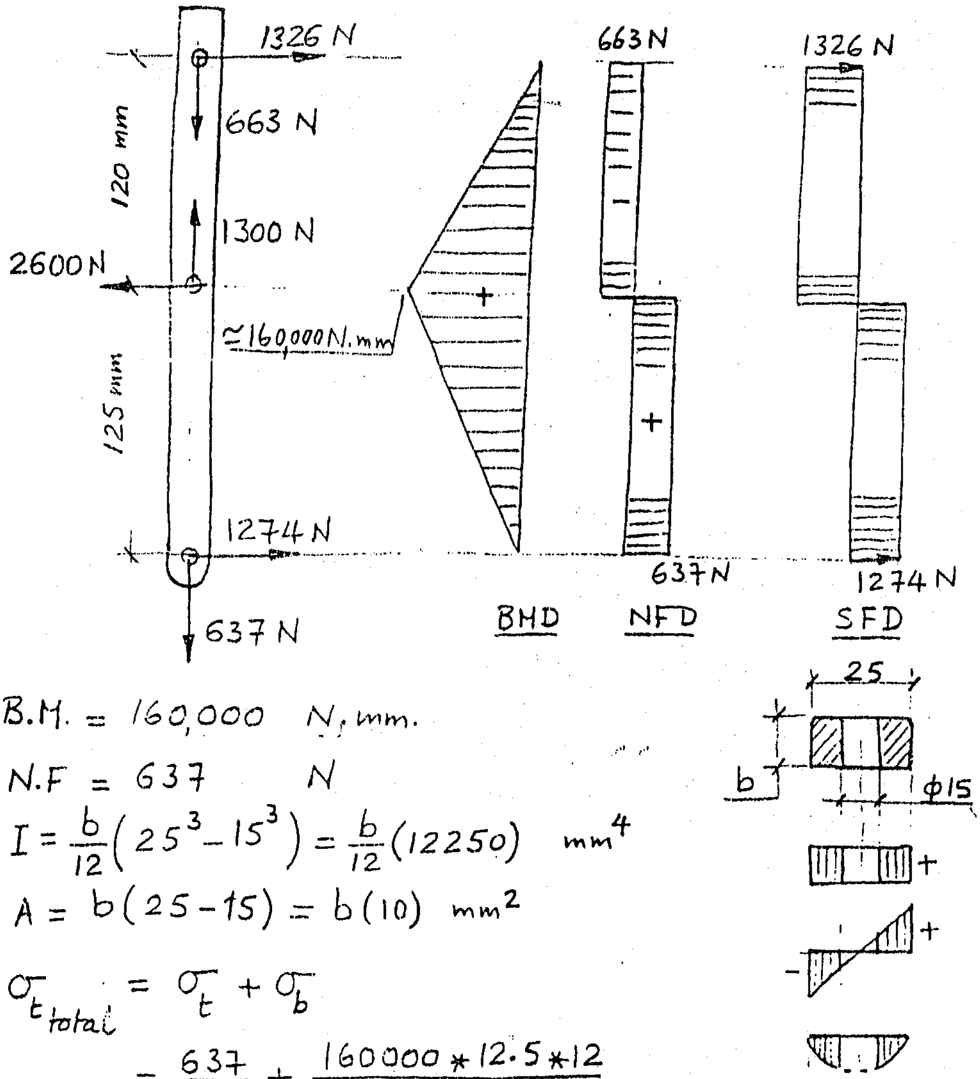
$b = 30$ mm

Note:

The normal force of 1379 N results in compression which reduces the tensile stress

The shear stress has a max. value at the neutral axis of the section where the bending stress is zero.

2) Left-hand lever :



B.M. = 160,000 N.mm.

N.F = 637 N

$I = \frac{b}{12}\left(25^3 - 15^3\right) = \frac{b}{12}(12250)$ mm^4

$A = b(25-15) = b(10)$ mm^2

$\sigma_{t_{total}} = \sigma_t + \sigma_b$

$= \frac{637}{10b} + \frac{160000 * 12.5 * 12}{12250\ b}$

$= \frac{63.7}{b} + \frac{1959}{b} = \frac{2023}{b}$

$\sigma_{t_{design}} = \frac{300}{4} = 75$ N/mm^2

$\therefore \frac{2023}{b} = 75$ i.e. b = 27 mm

Unification of as forged levers is a required.

$\therefore$ b is taken 30 mm

$\therefore \sigma_{t_{total\ working}} = \frac{2023}{30} = 67.4$ N/mm^2 — $\sigma_t = 67.4$ N/mm^2

$\therefore$ Working safety factor $= \frac{300}{67.4} = 4.45$ — Working S.F. = 4.45

3) Actuating Lever :

$P_{act} = 300$ N

$F_s = 900$ N

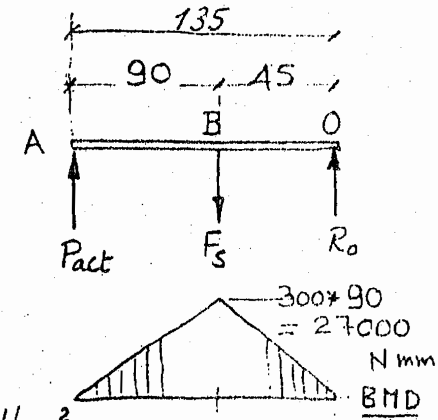
$R_o = 600$ N

Lever thickness = 12 mm

Material : St C15

$\sigma_{y_t} = 300$ N/mm²

SF. = 4 i.e $\sigma_{t_{design}} = 75$ N/mm²



Material :
C15
$\sigma_y = 300$ N/mm²

$\sigma_{t_{design}} = \frac{M}{Z}$

$\therefore Z \geq \frac{27000}{75} \geq 360$ mm³

$Z = \Sigma \frac{bh_i^2}{6} = \frac{10\,h^2}{6} - \frac{12\,d^2}{6}$, $d = 12$ mm

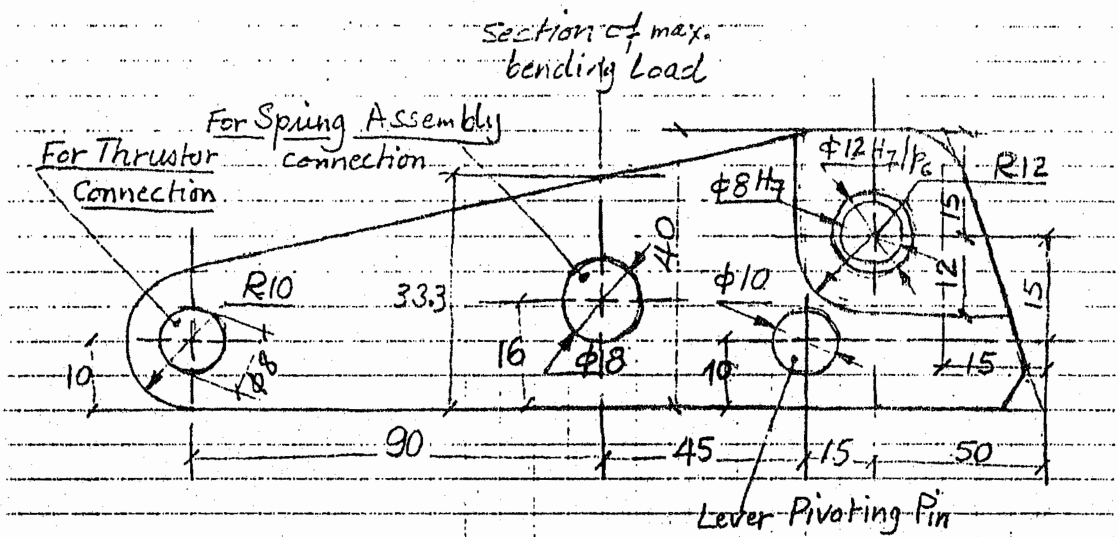
$\therefore 360 = 1.67(h^2) - 2(12^2) = 1.67h^2 - 288$

$h = \sqrt{(360+288)/1.67} = 20$ mm

take $h \geq 24$ mm with a hole $\phi 12$ about the neutral axis of the section in plane at B.

$Z \geq 360$ mm³

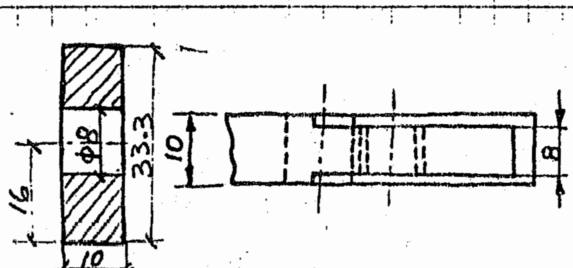
$\therefore h \geq 20$ mm



$Z_{actual} = \frac{10(33.3)^2}{6} - \frac{10(8)^2}{6}$
$= 1740$ mm³

$\sigma_{working} = \frac{27000}{1740} = 15.5$ N/mm²

Working margin of safety = 19



$Z_{act.} = 1740$ mm³

Working margin of Safety = 19
(rigid enough.

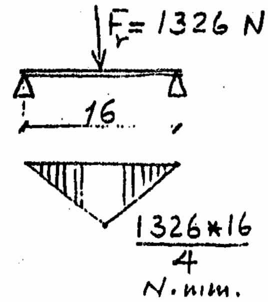
- **Bending stress on hinge pin:**

d = Pin diam. = 8 mm

$Z = \frac{\pi d^3}{32} = 50.3 \text{ mm}^3$

$M_b = 5304$ N.mm

$\sigma_b = \frac{5304}{50.3} = 105.5 \text{ N/mm}^2$



$\sigma_y \geqslant \sigma_b$ * margin of safety

margin of safety = 4

$\therefore \sigma_y \geqslant 105.5 * 4 = 422 \text{ N/mm}^2$

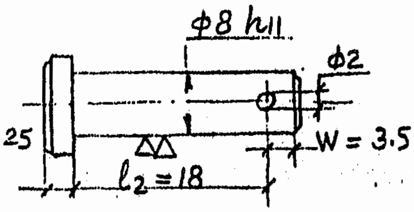
Pin material is st C60, $\sigma_y = 490 \text{ N/mm}^2$

Note:

Calculating the pin in this way is an extreme case. For St C45, $\sigma_y = 400 \text{ N/mm}^2$ is more common than st C60 but the yield stress (min. guaranteed) is 400 N/mm^2 which is less than the required value by 22 N/mm^2 = 5.5 % σ_y and can be used instead of St C60.

$\therefore$ Hinge pin material is St C45

- DIN 1444, Form B
- Ordinary washer DIN125 $D = 18$, $d = 9.5$ and $t = 2$ mm
- Split pin: DIN 935, $d \times \ell = 2 * 18$ mm



- **Bearing pressure on hinge pin:** [L.H. Lever]

Pin / actuating Lever, Bearing area = $\ell d = 8 * 8 = 64 \text{ mm}^2$

Bearing pressure = $\frac{1326}{64} = 20.7 \text{ N/mm}^2$

The allowable bearing pressure for Br/st = 17 N/mm^2

i.e. Using a bronze bush is a must with limited, but reasonable, life since the motion is limited.

$d = 8$ mm
$\ell = 16$ mm

$\sigma_{b\,working} = 105.5 \text{ N/mm}^2$

Material:
st C 60
$\sigma_y = 490 \text{ N/mm}^2$
and replaced by
St C 45
$\sigma_y = 400 \text{ N/mm}^2$

- Std. hinge Pin, DIN 1444 Form B
- Washer, DIN 125, $D = 18$ mm
- Split Pin, DIN 935 $\phi 2 * 18$

Bearing Pressure $\cong 21 \text{ N/mm}^2$ = 21 MPa
St/Bronz bush

4) Tie Rod : (Coupler)

Thread size $M10 \times 1.25$, core area $= 61.2 \ mm^2$

The min. section is the core area.

Force $= F_r = 1326 \ N$

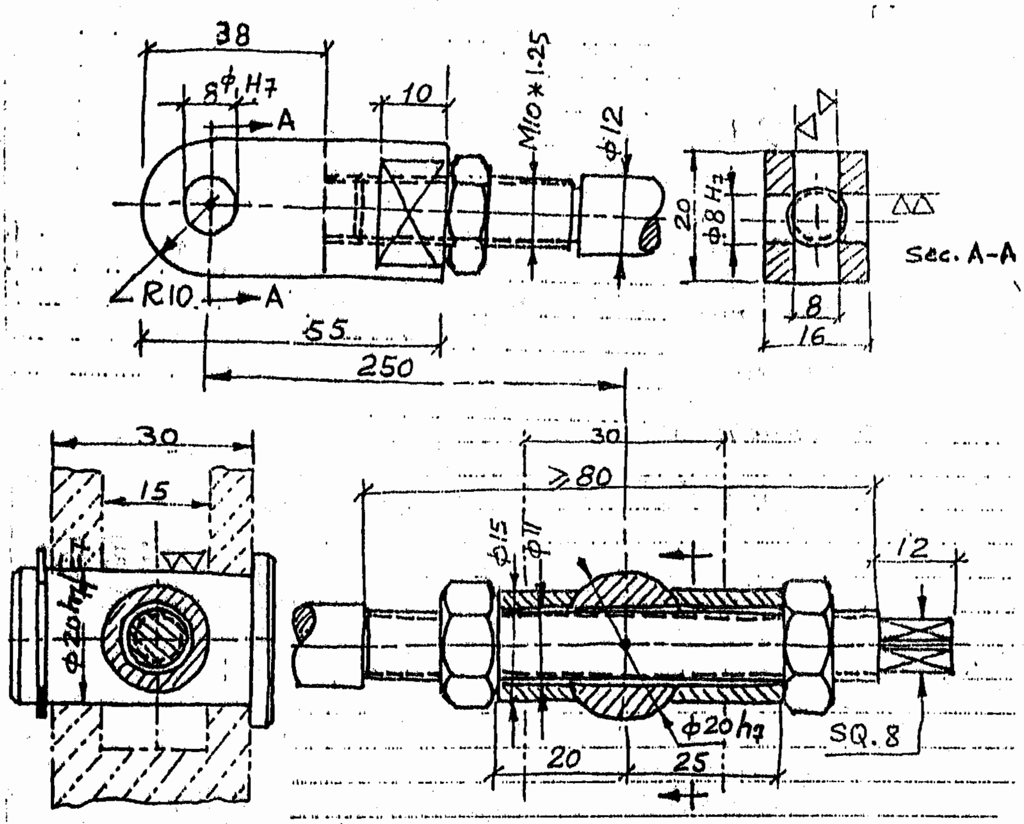
S.F. $= 4$

$$\text{Actual stress} = \frac{1326}{61.2} = 21.7 \ N/mm^2$$

$\therefore$ The material should have $\sigma_y \geq 21.7 * 4 \geq 87 \ N/mm^2$

For st 34, $\sigma_y = 190 \ N/mm^2$

$\therefore$ The coupler material is st 34



- Tensile stress on yoke eye:

area of section A-A $= 4(4*6) = 96 \ mm^2$

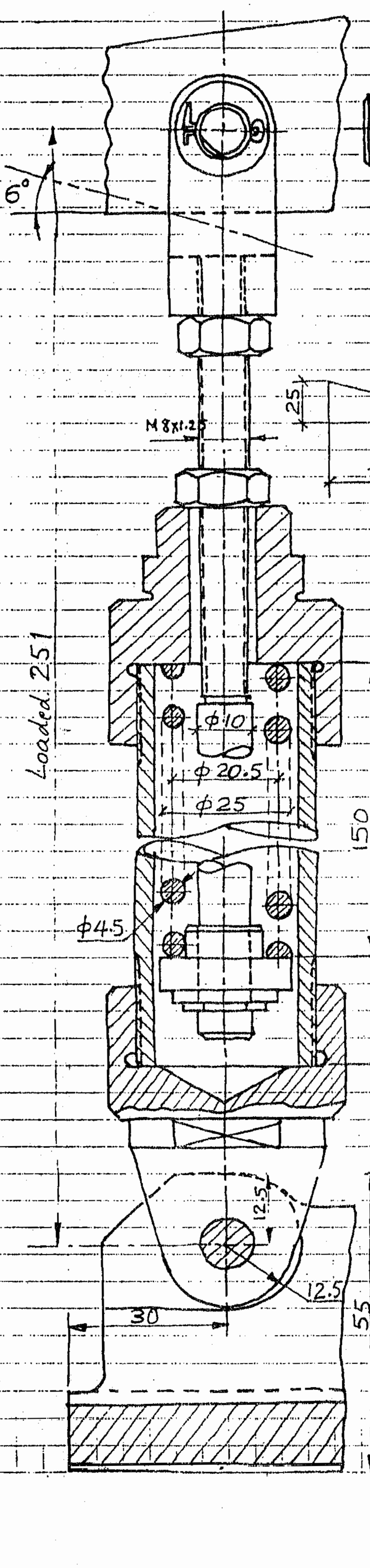
$$\text{tensile stress} = \frac{1326}{96} = 13.8 \ N/mm^2$$

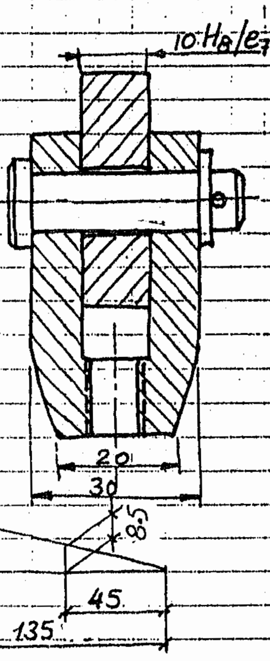
Suitable material is st 34, $\sigma_y = 190 \ N/mm^2$

$M10 * 1.25 \ mm$
Core Area $= 61.2 \ mm^2$

$\sigma_{working} = 21.7 \ N/mm^2$
Material: st 34, $\sigma_y = 190 \ N/mm^2$

$\sigma_{working} \cong 14 \ N/mm^2$
Material: st 34
$\sigma_y = 190 \ N/mm^2$





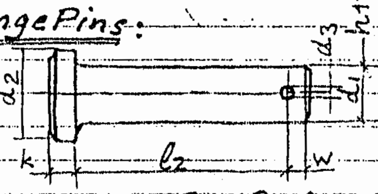
Pipes : (ASA Stds.)

Nominal	d"	D"	TPI/L₁"
1	1.049 (26.6)	1.315 (33.4)	$11\frac{1}{2}$/0.4 (≈9)
$1\frac{1}{4}$	1.380 (33.02)	1.66 (42.16)	$11\frac{1}{2}$/0.42 (10)
$1\frac{1}{2}$	1.610 (40.9)	1.900 (48.26)	$11\frac{1}{2}$/0.42 (11)
2	2.067 (52.5)	2.375 (60.33)	$11\frac{1}{2}$/0.43 (12)
$2\frac{1}{2}$	2.469 (62.71)	2.875 (73.03)	8/0.682 (18)
3	3.068 (77.93)	3.5 (88.9)	8/0.766 (20)

T.P.I. Threads per Inch.

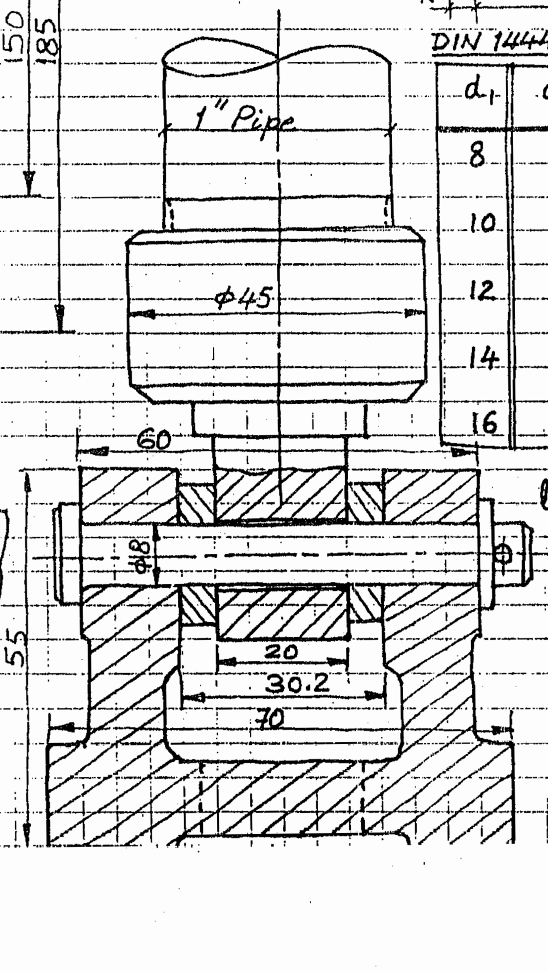
L_1" Length of engaged threads.

Hinge Pins :

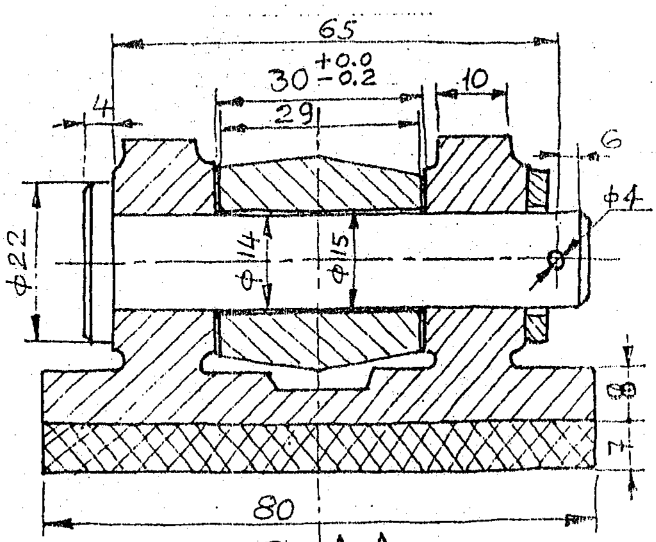


DIN 1444, Form B, Mar. 1974.

d_1	d_2	d_3	w	k
8	14	2	3.5	3
10	18	3.2	4.5	4
12	20	3.2	5.5	4
14	22	4	6	4
16	25	4	6	4.5



l_2 = 18, 20, (22), 25, (28), 30 100
5 to 5



Sec A-A

Selection of Lining material :

$$\text{Lining area} = \frac{\pi(20)*75^\circ}{360^\circ} * 8 = 105 \ cm^2$$

$$\text{Pressure intensity} = 3084/105 \cong 30 \ N/cm^2 = 300 \ kPa$$
$$= (43 \ psi) = (3 \ kg/cm^2)$$

Lining :
Max. pressure
= 300 kPa
= 0.3 MPa

* Some Properties of brake lining*

Property	Woven lining	Molded Lining	Rigid block
Compressive Strength (MPa)	70-100	10-18	10-15
Tensile strength (MPa)	17-21	27-35	21-27
Max. temperature (°C)	200-260	260	400
Max Speed (m/s)	38	25	38
Max. pressure (kPa)	340-690	690	1000
μ average (±10%)	0.45	0.47	0.4-0.45

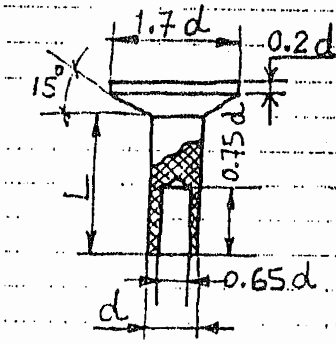
Molded lining is selected ($p = 690$ kPa) to achieve a longer service life. Drum max. rpm = 2300 :

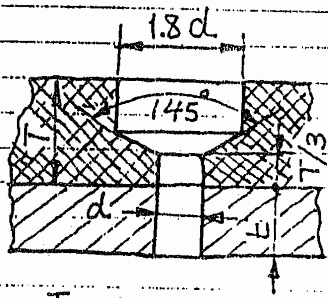
Following the brake block, the rivets in common use & Rivet diameter of 5 mm is selected.

Molded Lining,
$p_{max\,allowable} = 690$ kPa

Rivet :
ϕ5 mm.

* Mech.l Eng.g Design, J. E. Shigley, Mc Graw-Hill, 1989

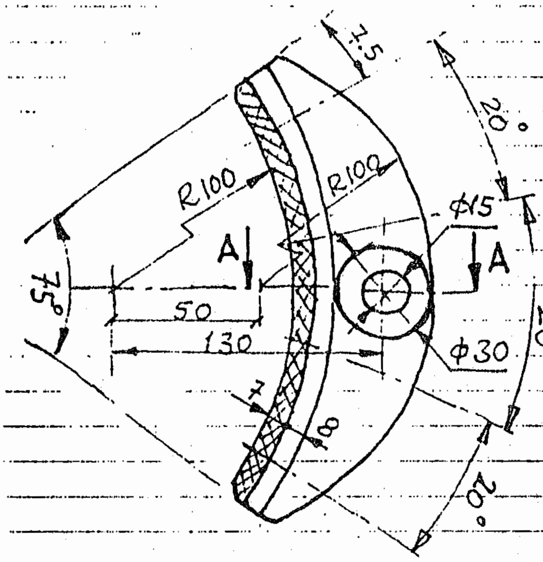


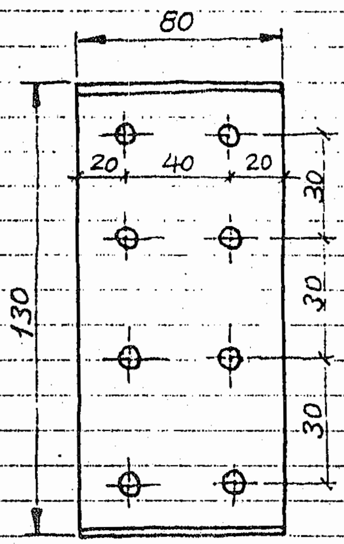


$L = \frac{T}{3} + t + d$

Code	D*	E	F	G	H	J
d (mm) metric	2.5	3	3.5	4	4.5	5
d (inch) imperial	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{9}{64}$	$\frac{5}{32}$	$\frac{11}{64}$	$\frac{3}{16}$
* Solid, Flat headed rivet						
max. Area/rivet cm^2	4	5.5	7.5	9.5	12	15
T (mm)	5	7	10	11	12	13

$T = 7$ mm



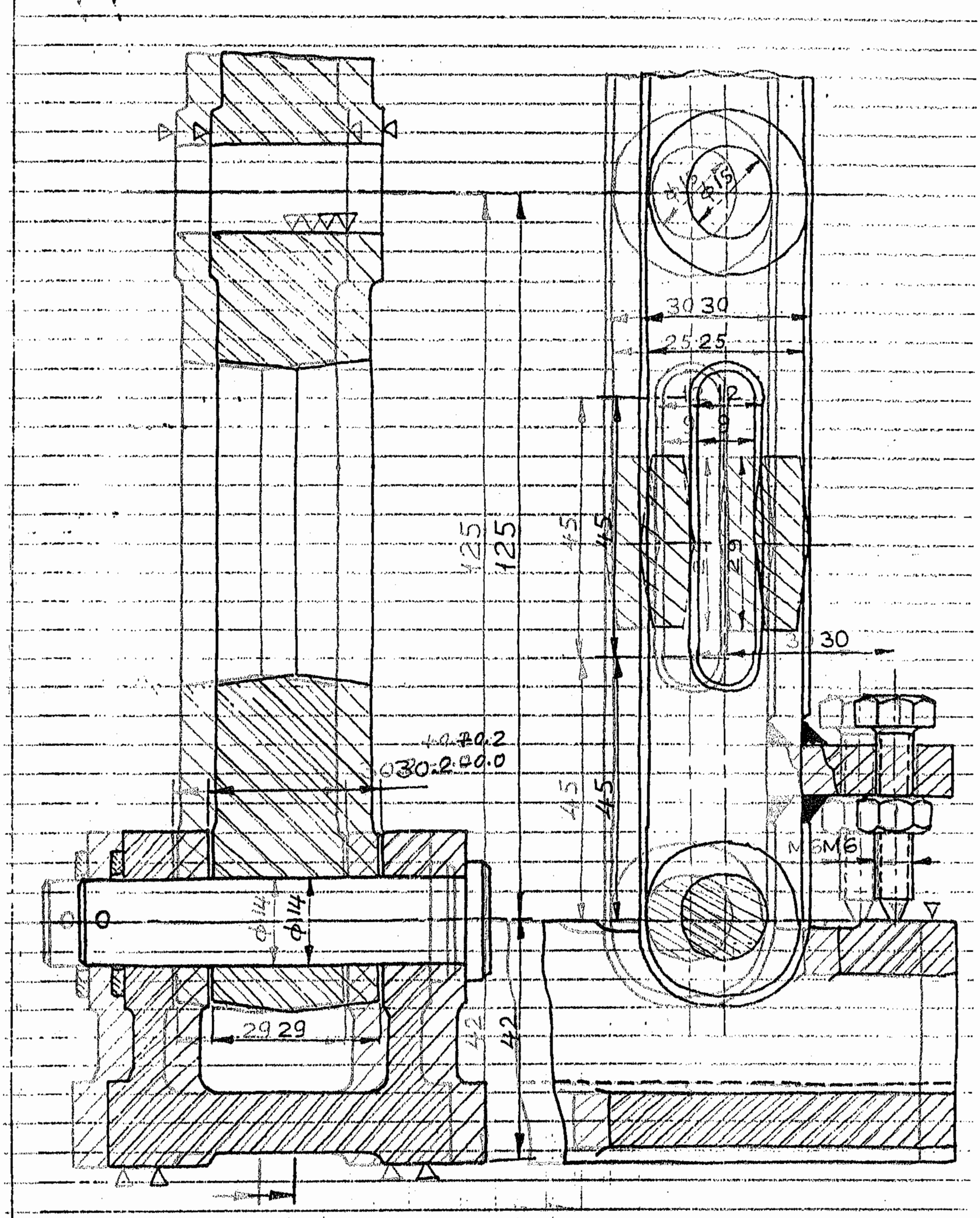


$$\text{Number of rivets} = \frac{\text{lining area}}{\text{area per rivet}} = \frac{13 * 8}{15} \cong 8 \text{ rivets}$$

$$L = \frac{T}{3} + t + d = \frac{7}{3} + 8 + 5 = 15 \text{ mm}$$

<u>No of rivets</u>
= 8 rivets
$\phi 5$, $L = 15$ mm

D-D4/49/19



125
125
30 30
25 25
12 12
9 9
45
45
29
30 30
45
45
M6 M6
ϕ30
ϕ14
ϕ14
29 29
42
42

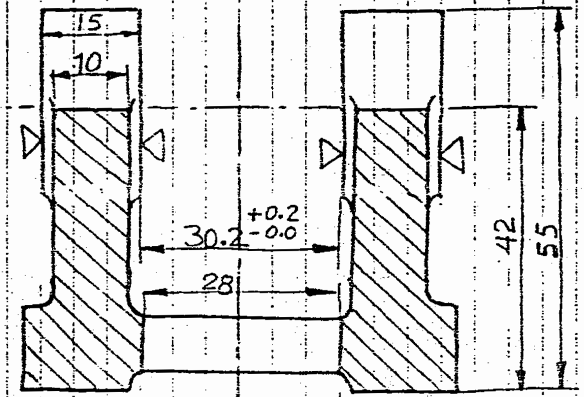
Aproximately:

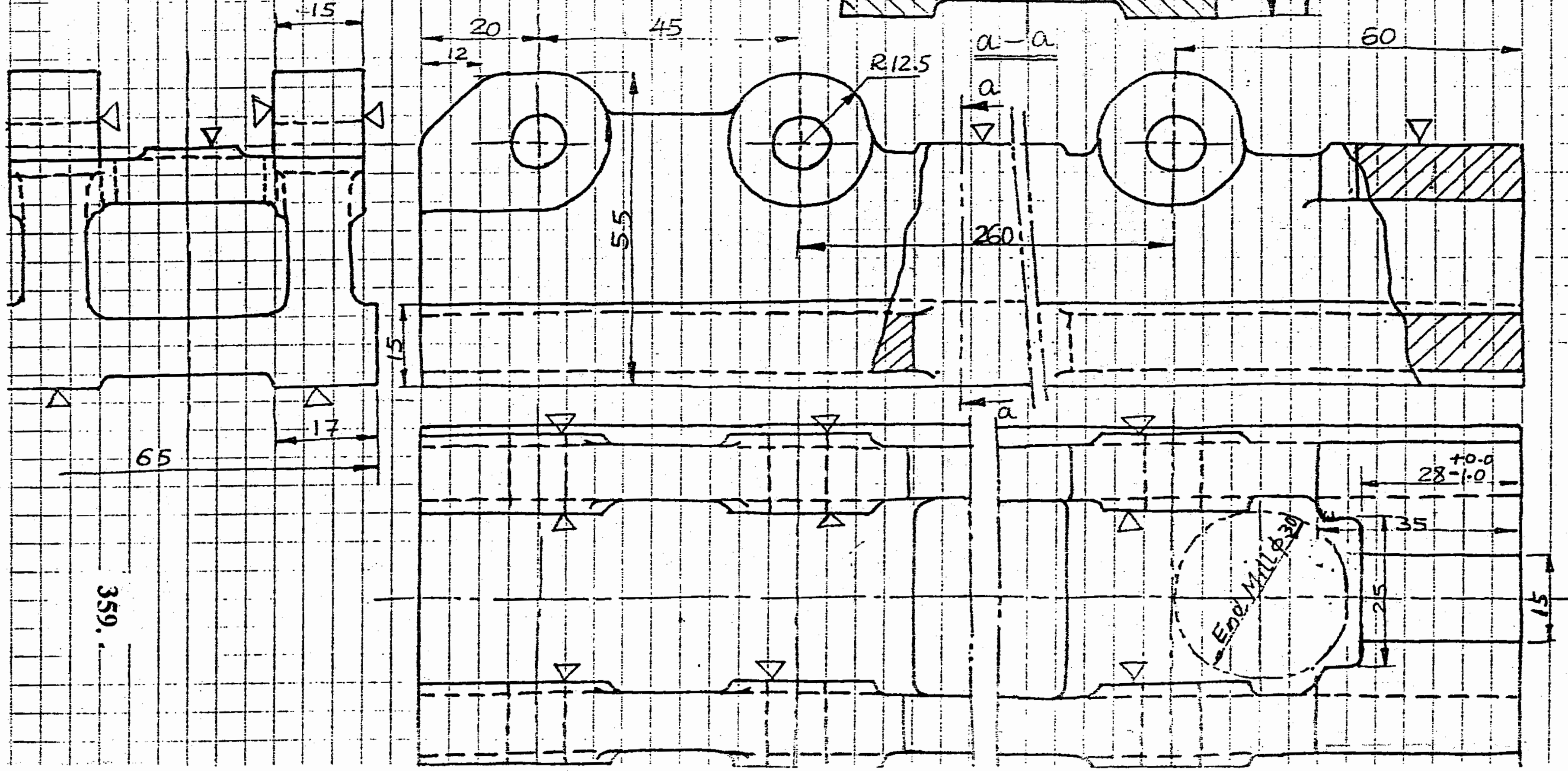
(For sec. a-a) $I = 2\left[\frac{10 \cdot (4.2)^3}{12}\right] = 12.35\ cm^4$

$y = 2.1\ cm$

$Z_b = \frac{I}{y} = 5.88\ cm^3$

$Area = 8.4\ cm^2$





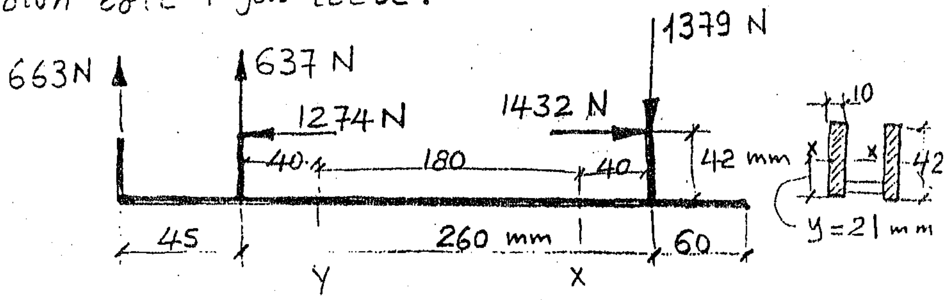
6) R.H.(and L.H) lever / base hinge pin Connection :-

Resultant force = $\sqrt{(1432)^2 + (1379)^2} = 1988$ N

This force is less than $R = 3084$ N which is the resultant force on the hinge pin of the brake shoe. For the sake of interchangeability, the same pin of the brake shoe; in size and material, is used with a higher working margin of safety

7) Brake base :

The loading diagram when the holding down bolt Y gets loose:



$Z_{xx} = 5880\ mm^3$

$A = 840\ mm^2$

$\Sigma Mts @ X = 0$

$M = [663 * (45 + 40 + 180)] + [637 * (40 + 180)] - [1274 * 42]$

$= 175{,}695 + 140{,}140 - 53508 = 262{,}327$ N. mm.

Stressed section properties :

$Z = 5880\ mm^3$, $A = 840\ mm^2$

$$\sigma_{norm.} = \frac{M}{Z} + \frac{F}{A}$$

$$= \frac{262327}{5880} + \frac{1274}{840} = 44.6 + 1.5 = 46\ N/mm^2$$

$\sigma_{max\ working} = 46\ N/mm^2$

$\sigma_y \geqslant 46 * \text{margin of safety} \geqslant 46 * 4 \geqslant 184\ N/mm^2$

The suitable material is cast steel GS 45, $\sigma_y = 220\ N/mm^2$

$\therefore$ Working margin of safety $= \frac{220}{46} = 4.78$

Material : GS 45 $\sigma_y = 220\ N/mm^2$

8) Holding down bolts:

Holding down force = 1962 N/bolt

Select the size and look for the suitable material.

"The bolt should be tightened to generate an axial load in excess of the force of 1962 N by about 150% that the other bolt will have $P_r = 50\% P_a$"

Axial load on the bolt ≅ 3000 N

Bolt size is M 14 * 1.5 , core area = 125 mm^2

σ_t = Normal Stress = $\frac{5000}{125} = 40$ N/mm^2

This is the working stress on the highly stressed bolt.

Calculation of the assembly stress

$$T = T_1 + T_2 = P_a \left[\frac{d_m}{2} \left(\frac{\tan\alpha + \mu'}{1 - \mu' \tan\alpha} \right) + \mu_s \frac{d_{ms}}{2} \right]$$

$$d_i = \sqrt{\frac{4*125}{\pi}} = 12.6 \text{ mm}$$

$$d_m = \frac{14 + 12.6}{2} = 13.3 \text{ mm}$$

$$\tan\alpha = \frac{p}{\pi d_m} = \frac{1.5}{\pi(13.3)} = 0.072$$

$$\mu_{st/st} = 0.25 \quad , \quad \mu'_{st/st} = \frac{0.25}{\cos 30^\circ} = 0.28$$

$$d_{ms} = \frac{22 + 16}{2} = 19 \text{ mm}$$

$$\therefore T = 5000 \left[\frac{13.3}{2} \left(\frac{0.072 + 0.28}{1 - 0.078 * 0.28} \right) + 0.25 \left(\frac{19}{2} \right) \right]$$

$$= 5000 [2.341 + 2.375] = 23580 \quad \text{N.mm}$$

$$\tau = \frac{16 * 5000 * 2.341}{\pi (12.6)^3} = 29.8 \quad \text{N/mm}^2$$

$$\tau_{comb} = \frac{1}{2} \sqrt{(40)^2 + 4(29.8)^2} = 35.9 \quad \text{N/mm}^2$$

$$\tau_y \geqslant 35.9 * 1.3 \geqslant 47 \quad \text{N/mm}^2$$

$$\sigma_y \geqslant \tau_y / 0.6 \geqslant 78 \quad \text{N/mm}^2$$

$\sigma_{y_{8.8}} = 64$ N/mm^2 , $\sigma_{y_{9.8}} = 72$ N/mm^2

St high grade 10.9 is selected, $\sigma_{y_{10.9}} = 90$ N/mm^2

Working margin of safety = $\frac{90}{40} = 2.25$, Margin$_{assembly}$ = 90/72 = 1.25

High Tensile Bolt
M14*1.5
Core Area = 125 mm^2

$\sigma_t = 40$ N/mm^2

Tightening Torque
T = 23.6 N.m.

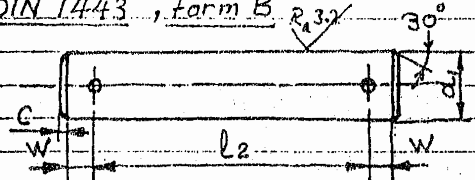
$\tau \cong 30$ N/mm^2

$\tau_{comb} \cong 36$ N/mm^2

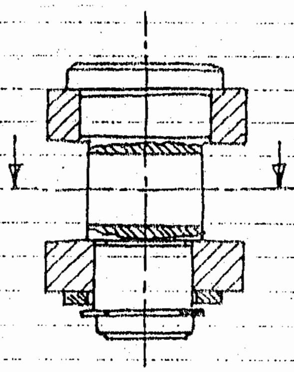
$\sigma_y \geqslant 78$ N/mm^2

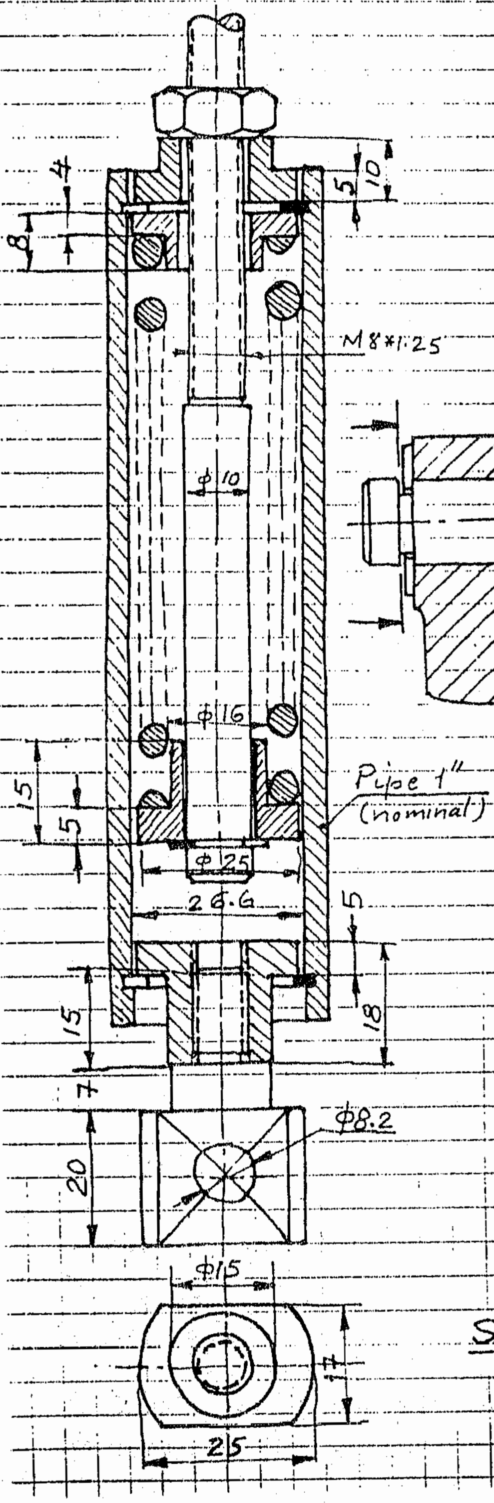
Material
St. Grade 10.9
Margines:
Working = 2.25
Assembly = 1.25

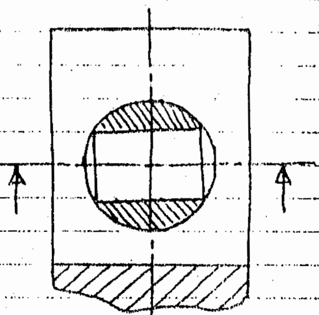
(The same is DIN EN 22340)
DIN 1443, Form B

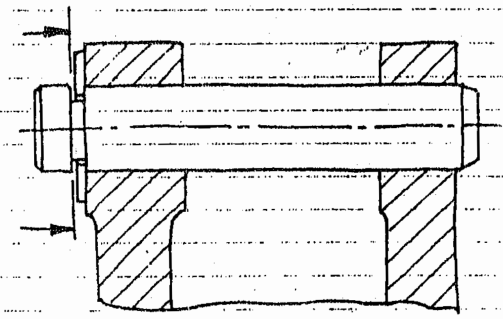


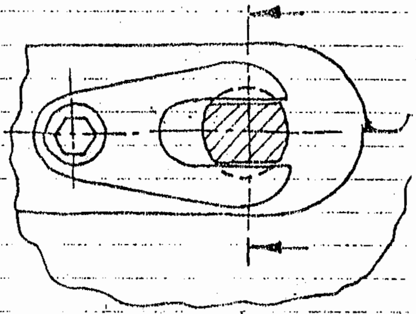
Form A: The same but without split pin holes, length $l_1 = l_2 + 2w$

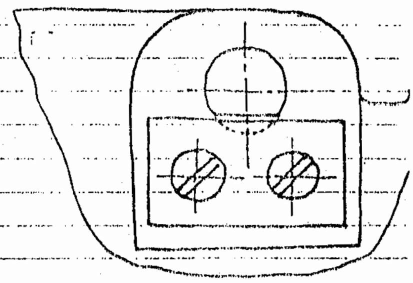












Some alternative joints and assemblies